Determination of Film Coefficients of Heat

Transfer of Crude Turpentine Gum

A Thesis

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Laurence Frederic Camp

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Abstract

Film-coefficients of heat transfer of a very viscous turpentine gum were determined, using a double-pipe, steam-jacketed heat exchanger. Two series of runs were made; one series of twenty-seven at a steam pressure of 5 lbs./sq. in. gage, and another series of twenty-one at 10 lbs./sq. in. gage.

Values for h, the film-coefficient, varied from 4.3 Btu/hr./sq. ft./°F to 49.9 Btu/hr./sq. ft./°F.

Mass velocity of fluid flow varied from 5.55 lbs./sq. ft./sec. to 97.80 lbs./sq. ft./sec. The region of turbulent flow was never reached, as the highest value of Reynold's Number, $\frac{du\rho}{\mu}$, obtained was 77.5.

The data is best correlated by plotting, on ordinary rectangular coordinates, values of h as ordinates and values of $(u\rho)\frac{(t_2-t_1)}{(T-t_1)}$ as abscissae. (For meaning of symbols see Table X.)

The equation for calculating the film-coefficient is,

$$h = 6.41\ (u\rho)\frac{(t_2-t_1)}{(T-t_1)}$$

Introduction

The subject of heat transmission has attracted the interest of many workers, primarily because of its fundamental importance in the design of condensers, heaters, and similar equipment. Excellent data are available in the literature of heat transmission for water, flowing in turbulent motion in clean pipes. McAdams and Frost (1), in a 1924 paper summarized the data of Stanton, Webster, and McAdams and Frost. The [illegible] by them as best correlating the then [illegible]:

$$\frac{hd}{K} = 0.0272\left(1+\frac{50}{r}\right)\left(\frac{du\rho}{\mu}\right)^{.8}$$

h - water film coefficient, Btu/hr./sq. ft./°F

d - inside diameter of pipe, feet

[illegible] conductivity of water, taken as 0.35

r - ratio of length of pipe to diameter.

$u\rho$ - mass velocity, lbs./ sq. ft./sec.

μ - absolute viscosity of water at the temperature of the film, lbs/(ft.)(sec.)

Their equation contains the factor, r, as a correction for tube length. Later work carried out by Lawrence & Sherwood (2) showed that tube length

(1) McAdams & Frost; Refrig. Eng.; 10; 323 (1924)
(2) Lawrence & Sherwood; Ind. & Eng., Chem.; 23; 301 (1931)

was apparently without effect on values of h. They used four pipes varying in length from 2.9 ft. to 11.1 feet in which the ratio of heated length to diameter varied from 59 to 225. The experiments were carried out using a clean copper tube, 0.593 inches inside diameter, surrounded by a steam-jacket. Thermocouples were used to determine the pipe surface temperature. The results plotted, as by McAdams & Frost, $\frac{hd}{K}$ vs $\frac{dus}{Z\,m}$, for all four pipes fall in a narrow band, but not in order of the pipe lengths. Thus, they concluded that the effect of pipe length on the film coefficient of heat transmission is negligible.

A graphical analysis of the data on the over-all coefficients of heat flow from steam to water, not requiring the use of thermocouple readings, led to the same conclusion.

Eagle and Ferguson (3) give some valuable data on coefficients of heat transfer from tube to water, but do not agree exactly with some other investigators as to the net exponent on the tube diameter, d. They agree, however, that h is an inverse function of d.

(3) Eagle & Ferguson; Engineering; 130; 691, 788, 821 (1930)

The classical work of Orrak (4) on the general subject of heat transmission in surface condensers is a model of accuracy and reliability. It has the disadvantage; however, of dealing with over-all coefficients rather than film coefficients, because it was done before the present practice of determining film coefficients was begun.

Heat transmission from metal surfaces to boiling water has received little attention. Cryder and Gilliland (5) have published some data on boiling water. Their values for h ranged from 380 to 4000. Their apparatus will be described later.

Linden and Montillon (6) working with a small inclined-tube evaporator, calculated the boiling-liquid film coefficients of water inside a copper pipe from direct measurements of the temperatures of the outer surface of the pipe and of the main body of the liquid. The temperature drop through the pipe wall was estimated from the known value of thermal conductivity for copper. Their values for h ranged from 250 to 1500.

(4) Orrak; Trans., A.S.M.E.; 1139 (1910)
(5) Cryder & Gilliland; Ind. & Eng. Chem; 24; 1382 (1932)
(6) Linden & Montillon; Ind. & Eng. Chem; 22; 708 (1930)

Jakob and Fritz (7) measured the boiling liquid film coefficients of water, using electrically heated horizontal metal plates instead of a tube or pipe. Polished, sanded and grooved surfaces were employed. As might be expected, the sanded and grooved surfaces gave the highest values for the coefficient of heat transmission, since they provided the greatest number of nuclei for bubble formation. Values for h for the grooved surface ranged from 800 to 3000, for the sanded surface, from 200 to 1200, and for the polished chromium-plated surface, they ranged from 200 to 800.

Heat transmission data for liquids other than water are rather meager. Cryder and Gilliland (5), Sherwood and Petrie (8), Morris and Whitman (9) and Garcia (10) have obtained data on film coefficients for various liquids in turbulent flow.

Cryder and Gilliland investigated ten liquids by means of an experimental evaporator consisting of an electrically heated brass tube suspended in the liquid. Suitable thermocouples were used to obtain the temperatures of both the pipe and liquid of widely

(7) Jakob & Fritz; Forsch. Gebiete Ingenievrw; 2; 453 (1931)
(8) Sherwood & Petrie; Ind. & Eng. Chem.; 24; 736 (1932)
(9) Morris & Whtiman; Ibid.; 20; 234 (1928)
(10) Garcia; Ibid.: 20; 889 (1928)

varying temperatures differences for each of the ten liquids. The liquids used and the extremes of the valves of h are given in Table I.

Table I

Liquids Investigated and Extremes of Values Obtained by Cryder and Gilliland

Liquid	h : Btu/hr/sq. ft./°F Low	High
9.6% Na_2SO_4 solution	370	2800
9.1% NaCl solution	320	2700
26% Glycerol solution	330	2800
25% Sucrose solution	240	1800
26% NaCl solution	200	1400
Methonol	100	1250
Kerosene	140	930
Gasoline	120	950
Carbon Tetrachloride	120	920
N-Butanol	100	850

Sherwood and Petrie (8) worked with acetone, benzene, kerosene, and n-butanol flowing in both streamline and turbulent motion through a 0.494 inch inside diameter steam-jacketed copper pipe. The data obtained in turbulent flow were well correlated by the Dittus and Boelter (11) equation:

$$\frac{hd}{K} = 0.024 \left(\frac{du\rho}{\mu}\right)^{.8} \left(\frac{c\mu}{k}\right)^{.4}$$

(11) Dittus & Boelter: Univ. Calif. Pub. Eng.; Bull. 2 (1930)

h - film coefficient

d - tube diameter, feet

u - mean fluid velocity, ft./sec.

k - thermal conductivity)	Taken at the main body average temperature. Any set of consistent English units may be used.
μ - absolute viscosity)	
ρ - fluid density)	
c - specific heat)	

Table II shows the liquids studies and the extremes of the valves of h obtained.

Table II

Liquids Investigated and Extremes of Values of h
Obtained by Sherwood and Petrie

	h : Btu/hr./sq. ft./°F	
Liquid	Low	High
Acetone	19.3	1250
Benzene	7.4	1005
Kerosene	13.5	804
N-Butanol	37.0	562

The most important data on petroleum oils are those of Morris and Whitman (9). They determined the film coefficients of heat transfer for three oils flowing through the steam-surrounded tube of a double-pipe heat exchanger. Flow was turbulent or semiturbulent; linear velocity varied from 1 to 20 feet per second; and viscosity changed from 0.5 to 55 centipoises. The values of the film coefficients covered a range from 10 to 700 Btu/hr./sq. ft./°F. The lower values were obtained, of course, in the region of semiturbulent flow. Physical properties of the oils were taken at the temperature of the main body of the liquid.

In the region of viscous or stream-line flow the published data consists almost entirely of the work of Drew, Hogan, and McAdams (12), Drew (13), McCormick and Diederichs (14), and Kirkbride and McCabe (15).

Drew, Hogan, and McAdams collected data from the heating of a hydrocorbon oil flowing inside a steam-heated horizontal copper pipe having an inside diameter of 0.5 inch and a heated length of approximately 5 feet. The Reynold's Number, $\frac{du\rho}{\mu}$ in consistent units, varied from 2 to 1400. (Turbulent flow is not obtained until the value of $\frac{du\rho}{\mu}$ reaches 2000 to 2300).

Drew (13) reported new data on heat transfer to glycerol in stream-line flow through a horizontal 1/8 inch (Brigg's Standard) copper tube, steam-heated over 61.75 inches of its length. $\frac{du\rho}{\mu}$ ranged from

(12) Drew, Hogan, & McAdams; Ind. Eng. Chem.; 23; 936 (1931)
(13) Drew: Ibid.; 24; 152 (1932)
(14) McCormick & Diederichs; Cornell Univ.; Eng. Ex. Sta.; Bull 7 (1927)
(15) Kirkbride & McCabe; Ind. & Eng. Chem.; 23; 625 (1931)

7.2 to 109.2 and values for h varied from 24.8 to 66.9 Btu/hr./sq. ft./°F. No one equation that satisfactorily correlated the data was given.

A very viscous, heavy Mexican crude oil was used by McCormick and Diederichs (14) in their investigation. Commercial size apparatus was employed. Their weight rate of flow varied from 484.2 lbs oil/hr. to 2412 lbs./hr. That corresponded to a linear velocity range of from 1.01 ft./sec to 5.12 ft./sec. The values of h reported by them ranged from 18.4 to 86.2 Btu/hr./sq. ft./°F.

Kirkbride and McCabe (15) carried out experiments on two oils, one a light and the other a heavy fuel oil. The linear velocity was varied from 74.5 lbs./hr./sq. ft. to 4340 lbs./hr./sq. ft. Values obtained for h varied from 6.77 to 44.3 Btu/hr./sq. ft./°F. Their highest values of h was obtained when the linear velocity was only 575 lbs./hr./sq. ft. This fact indicates that in the region of viscous flow, values of h vary not only as the mass velocity, but also as the temperature rise of the fluid and the ratio of the temperature rise to the initial temperature difference between the pipe wall and the incoming fluid.

Scope of Investigation

As pointed out above heat transmission data on fluids in viscous flow are limited to a few petroleum oils and glycerol. This investigation was carried out in order to determine the surface film coefficients of heat transfer of a very viscous, crude turpentine gum in stream-line flow. The gum is the raw material from which turpentine and naval stores are obtained. Although the distillation of turpentine gum has been practiced in the naval stores region of the south for many years, the physical properties of the gum itself have received practically no attention.

The viscosity-temperature relation as determined by Houze (16) is given in Table III and Fig. I. The physical properities of the gum used in this work are shown in Table IV.

(16) Houze; G.S.T. Undergraduate Chem. Eng. Thesis (1933)

Table III

Viscosity-Temperature Relation of Crude Turpentine Gum

Temperature		Viscosity	
Degrees C	Degrees F	Poises	Absolute lbs./(ft.)(sec.)
30	86	45.100	3.03072
40	104	5.485	0.36859
50	122	1.293	0.08688
60	140	0.386	0.02394
70	158	0.117	0.00786
80	176	0.052	0.00349

Table IV

Physical Properities of Crude Turpentine Gum

Viscosity	0.880 lbs./(ft.)(sec.)	0.071 lbs./(ft.)(sec.)
Specific Heat	0.415 Btu/lb./°F	
Density	63.4 lbs./cu. ft.	

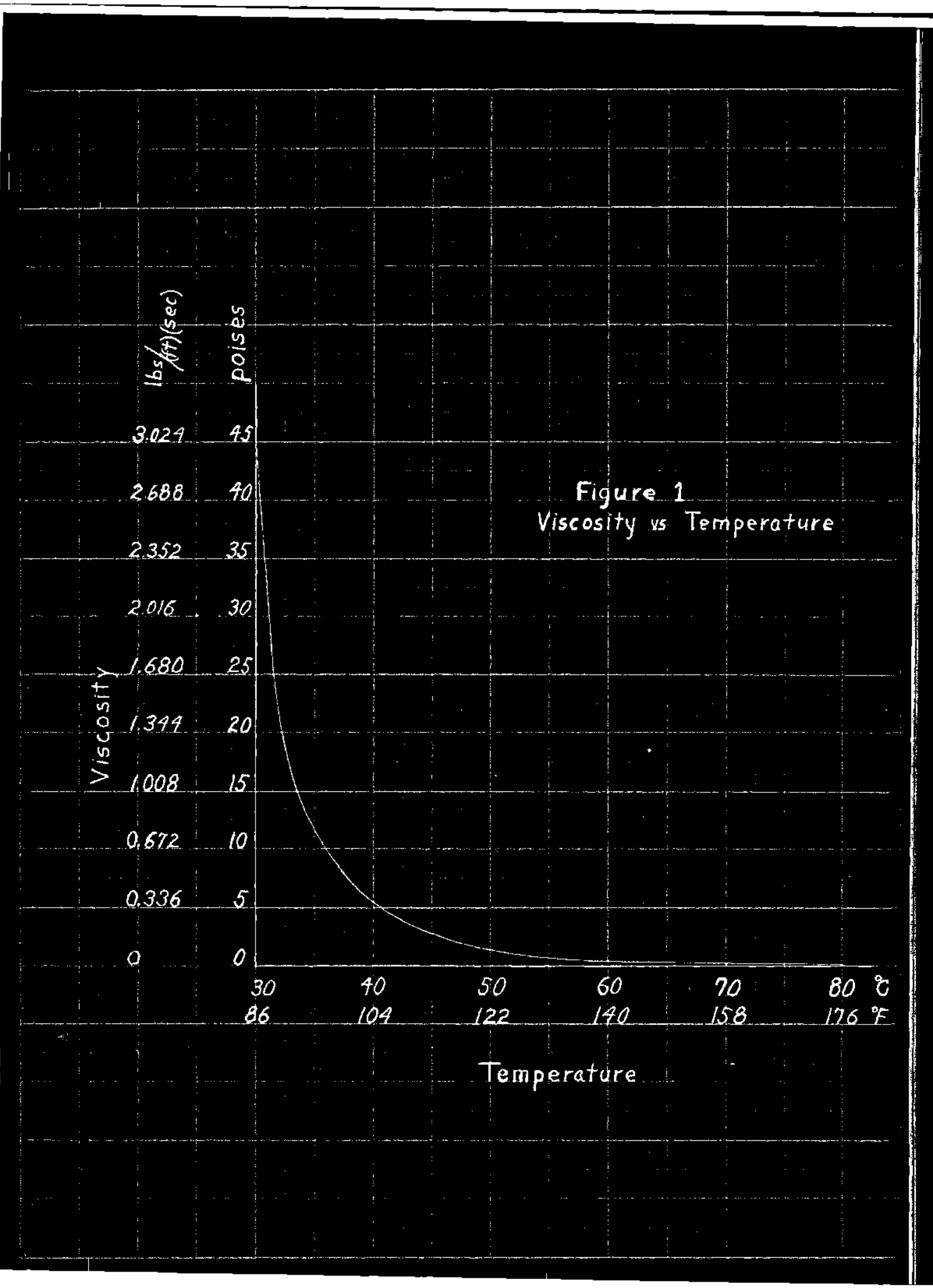
Figure 1
Viscosity vs Temperature
Viscosity
lbs/(ft)(sec)
poises
3.024 45
2.688 40
2.352 35
2.016 30
1.680 25
1.344 20
1.008 15
0.672 10
0.336 5
0 0
30 40 50 60 70 80 °C
86 104 122 140 158 176 °F
Temperature

Description of Apparatus

The apparatus consists of a double-pipe, steam-jacketed heat exchanger with the tanks, valves, pump and connections necessary for continuous recycling of the fluid. Figures 2a and 2b are photographs of the assembled apparatus. The heat-exchanger is designated by G. The weighing tank A was connected into the system with lengths of rubber hose approximately thirty inches long. B and B' are storage and supply tanks. C is a Gardner-Denver doubling-acting duplex steam pump. This type of pump was chosen because of its ability to handle the turpentine gum at relatively low temperatures where the viscosity is very high. The air chamber F was installed to insure uniform flow of the fluid. Outlet and inlet thermometer wells are indicated by M and P. N and O are thermometer wells in the steam space. Fluid velocity was regulated by the by-pass E, and by the rate of pumping. H is the condensate receiver, which was fitted with a standard water-level gauge. The gauge was necessary in order to keep the condensate level approximately constant.

Figure 2a

Figure 2b

A detailed view of the heat exchanger is shown in Figure 3. The two thermometer wells, N and O, reach well into the steam space. The other two thermometer wells, M and P, are in the fluid path and extend almost to the bottom of the tube. The exchanger is inclined slightly in order to insure complete drainage of the condensate. R and R' are "calming sections", each being twenty tube diameters in length. These calming sections practically eliminate end effects, (9). Consequently, uniform velocity conditions may be expected throughout the tube where heat is being transferred, and the calculated coefficients may be considered as applying to pipes of infinite length. The entire heat exchanger, including the calming sections and Y's holding the thermometer wells, was covered with Johns-Manville 85% Magnesia pipe covering. The dimensions of the heat exchanger are shown in Table V.

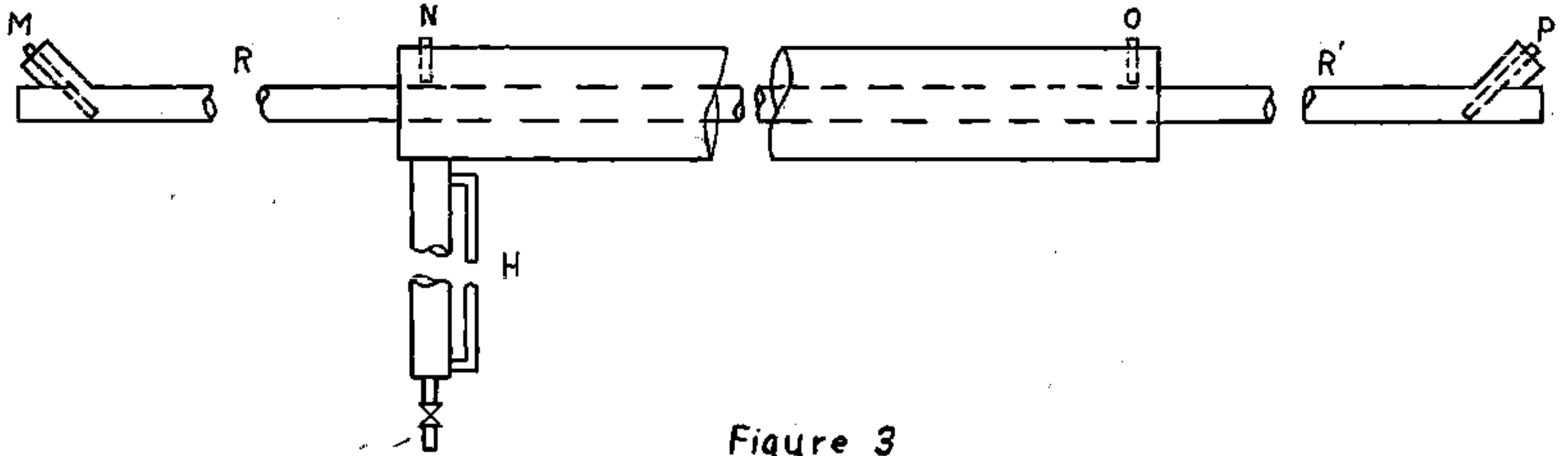

Figure 3
Detailed View of Heat Exchanger

Table V

Dimensions of the Heat Exchanger

Diameter of pipe, internal	1.049	inches
Diameter of pipe, external	1.315	inches
Cross section, internal	0.006	sq. ft.
Heated surface, length	68.25	inches
Heated surface, internal area	1.561	sq. ft.
Heated surface, external area	1.958	sq. ft.
Diameter of steam-jacket	3	inches
Length of each calming section	20	inches
Ratio, total length to inside diam.	107	

Experimental Proceedure

Data and Computations on Calibration of Apparatus

Several runs were made using water as the heated fluid. The object of these runs was to determine the steam-film heat transfer coefficient of the apparatus. The results of these runs are shown in Table VI.

The Btu's absorbed by the water were determined by multiplying the pounds of water pumped through the system in a definite time by the temperature rise as read from the thermometers located at M and P, and the specific heat of water (taken as a constant value of 1). Heat was supplied from steam sufficiently wet and at such a pressure (135 lbs./sq. in.) that it was dry, or contained only a degree or two of superheat, when reduced through the control valve to the pressure in the steam-jacket (2 to 5 lbs. gage).

The overall coefficient of heat transfer, U, was calculated from the heat absorbed by the water, Q, the temperature drop, ΔT, and the condensing area of the inner pipe, A, using the familiar form of Newton's Law

Table VI

Calibration Data Using Water as Heated Fluid

Run	t_1	t_2	T	θ	W	Q	ΔT	U	$\frac{1}{U}$	$\frac{1}{u^{.8}}$
B-8	53.4	117.3	215.6	3.0	52.8	67,350	130.3	264	0.00378	1.215
B-7	53.4	,116.4	215.6	4.0	73.0	69,000	130.7	270	0.00370	1.180
A-4	84.2	113.7	212.0	3.5	71.5	36,150	113.0	164	0.00610	1.080
B-6	54.3	102.7	214.0	3.5	86.0	71,390	135.5	269	0.00372	0.930
A-5	99.0	122.7	212.0	4.0	101.0	35,750	101.1	180	0.00555	0.911
C-5	61.2	98.9	217.4	4.0	102.5	57,800	137.4	215	0.00465	0.878
B-2	55.2	99.1	213.8	3.0	90.3	79,240	136.6	297	0.00337	0.793
B-4	55.0	98.4	221.9	3.0	91.3	86,820	145.2	305	0.00328	0.787
B-5	53.8	98.4	217.4	2.75	84.0	81,620	139.3	295	0.00339	0.715
B-1	54.7	87.6	213.4	3.0	151.0	99,360	142.6	356	0.00281	0.525
B-3	54.3	86.9	217.4	2.0	114.8	112,210	146.8	390	0.00256	0.472

$Q = U \Delta T A$

Q - Btu/hr.

U - Btu/hr./sq. ft. of condensing surface/°F temp. drop, steam to arithmetical mean water temperature.

ΔT - Temperature drop, steam to mean water temperature.

A - Area of condensing surface, 1.958 sq. ft.

The value of the steam-film coefficient, h_s, was obtained* from U by the method first used by E. E. Wilson (17) in a 1915 paper. It was later used by McAdams, Sherwood, and Turner, (18) in a comprehensive paper on heat transmission from condensing steam to water in surface condensers and feedwater heaters. McAdams (19) published a detailed description of the method in a 1927 paper.

The method is based on the wellknown principle that total resistance to heat flowing thru a series of resistances is equal to the sum of the individual resistances.

(17) Wilson; Trans. A.S.M.E. ; 37; 47 (1915)
(18) McAdams, Sherwood, & Turner; Ibid; 48; 1233 (1926)
(19) McAdams; Chem. & Met. Eng; 34; 599 (1927)

$$R = r_s + r_p + r_w$$

R - total resistance

r_s - resistance due to steam film

r_p - resistance due to pipe

r_w - resistance due to water film

Furthermore the over-all resistance, R, is equal to the reciprocal of the over-all coefficient;

$$R = \frac{1}{U}$$

and the resistance of the water film is inversely proportional to the linear velocity of the water.

$$r_w = \frac{1}{f(u)}$$

f(u) - some function of linear velocity.

It has been found by direct measurement (1) that f(u) can be taken as $b(u)^{.8}$, where b is an empirical constant and may be considered as the apparent individual coefficient of heat transfer from tube to water at a water velocity of 1 ft./sec. Substituting $\frac{1}{U}$ for R and $\frac{1}{b(u)^{.8}}$ for r_w gives

$$\frac{1}{U} = r_s + r_p + \frac{1}{b(u)^{.8}}$$

Except where very high water velocities are used the water-side resistance is usually the major resistance from condensing steam to water, and, under ordinary conditions, serious error would not be introduced by assuming that the sum of $r_s + r_p$ is approximately constant (20). Hence, a plot of $\frac{1}{U}$ vs $\frac{1}{(u)^{.8}}$ should give a straight line when plotted to ordinary rectangular coordinates. Such a plot for the data on water is shown in Fig. 4. When $u = 0$, $\frac{1}{U} = r_s + r_p$, or from Fig. 4, $r_s + r_p = 0.00086$, the intersection of the straight line and the ordinate.

Then $r_s = 0.00086 - r_p$

The value of r_p is readily calcualted from the known thermal conductivity of the pipe and the thickness of the pipe wall.

$$r_p = \frac{L}{k}\frac{(D')}{\left(\frac{D'+D}{2}\right)}$$

L thickness of pipe wall, 0.011 ft.

k thermol conductivity of pipe wall, 35 Btu/hr./sq. ft./°F/ (21)

D' outside pipe diameter 1.315 inches

D inside pipe diameter 1.049 inches

Substituting and solving for r_p

$$r_p = 0.000352$$

$$r_s = 0.000860 - 0.000352$$

$$r_s = 0.000508$$

$h_s = \frac{1}{r_s} = 1968$ Btu/hr./sq. ft./°F (steam-film coefficient)

(20) McAdams; Heat Transmission; McGraw-Hill, New York; 265 (1933)

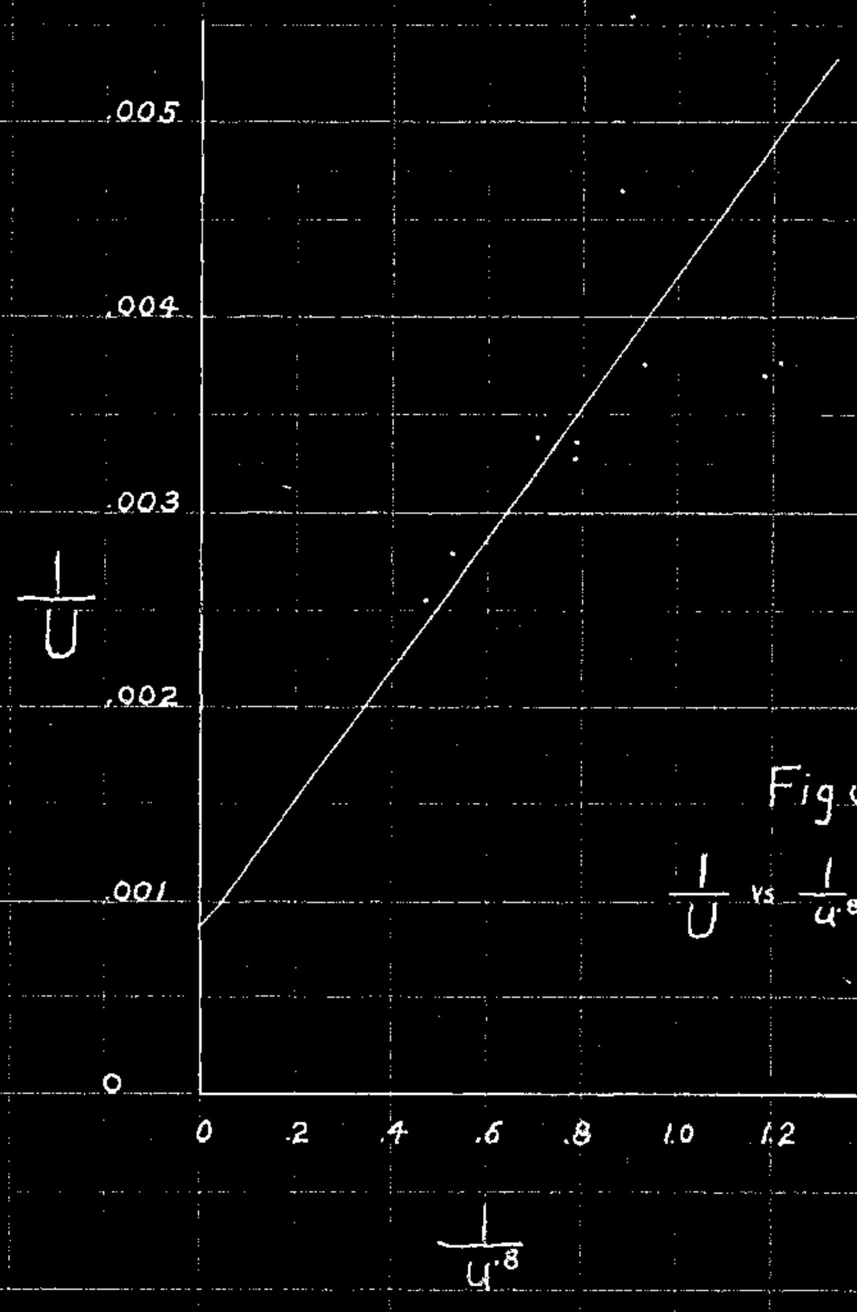

Figure 4

$\frac{1}{U}$ vs $\frac{1}{u^{.8}}$ for Water Data

This value of h_s corresponds well with published values of the steam film coefficients (22), (23). It is remarkably close to the value of 1950, published by Nusselt (24) in 1910 as an average value of h_s.

The slope of the line of Fig. 4 is the reciprocal of b, in $r_w = \frac{1}{b(u)^{.8}}$ or $b = 297$.

Therefore, the empirical equation for fluids in turbulent flow through the heat exchanger is;

$$\frac{1}{U} = 0.00086 + \frac{1}{297(u)^{.8}}$$

(21) Marks; Mech. Eng. Handbook, McGraw-Hill New York; *
(22) Othmer, Ind. & Eng. Chem; 21; 576 (1929)
(23) Clement and Garland; Univ. of Ill. Eng. Ex. Sta.; Bull. 40 (1909)
(24) Nusselt; Mitt. Forschungsarbeiten; No. 89 (1910)
*310 (1924)

Data and Computations Using Turpentine Gum

Two series of runs were made on the gum: Series D consists of 27 runs at a pressure in the steam jacket of 5lbs./sq. in. gage; Series E consists of 21 runs at 10 lbs./sq. in. gage. The data are shown in Tables VIII and IX. (For nomenclature see Table X.) The range of data is shown in Table VII.

Table VII

Range of Data

	Series D		Series E	
	Low	High	Low	High
Inlet Temperature, °F	85.0	118.2	86.6	119.2
Outlet Temperature, °F	105.3	138.0	106.4	131.1
Temperature rise, °F	5.9	45.6	4.7	39.3
Mass velocity, lbs./sq. ft./sec.	5.55	84.9	15.28	97.8
Velocity, ft./sec	0.087	1.34	0.24	1.54
Reynold's Number, $\frac{du\rho}{\mu}$	1.51	65.0	2.86	77.5

Table VIII

Run "D"	t_1	t_2	T	P	W	θ	w	q_s	q_g
1	108.1	121.5	223.7	5.0	48.5m	6	2.9	111	270
2	110.2	117.5	223.0	5.0	27.0	6	2.3	75	82
3	112.0	116.8	223.0	5.0	45.0	6	2.9	111	90
4	94.8	106.3	223.0	5.0	70.5	6	6.1	300	336
5	106.6	112.5	223.0	5.0	112.5	4	5.4	259	275
6	118.2	126.8	224.8	5.0	117.0	5	9.4	495	417
7	117.4	129.6	225.0	5.2	136.5	9	12.3	640	690
8	87.0	105.3	224.6	5.4	40.0	5	8.9	465	304
9	88.2	116.2	224.6	5.8	59.0	7	12.3	665	676
10	88.8	120.3	223.7	5.1	56.0	6	9.4	495	750
11	99.0	108.3	224.9	5.2	81.0	5	7.8	400	313
12	99.4	115.9	224.4	5.3	73.0	5	11.8	635	500
13	94.0	125.0	224.6	5.2	33.0	3	3.8	195	424
14	106.5	112.5	224.2	5.1	107.0	3.5	8.9	465	267
15	104.0	119.2	224.2	5.1	10.0	5	2.7	99	63
16	109.9	116.5	224.4	5.2	131.0	5	7.8	400	359
17	86.9	121.5	223.4	4.9	13.0	5	5.2	247	187
18	85.0	117.7	223.3	4.9	26.0	5	7.2	365	357
19	89.1	122.6	224.3	5.1	27.0	5	6.2	306	375
20	87.4	122.3	224.1	5.2	27.0	5	7.5	382	391
21	98.8	110.6	223.6	4.8	61.0	5	9.5	500	330
22	88.3	112.6	224.3	5.2	16.0	5	6.0	294	161
23	90.6	129.5	225.0	5.1	17.0	5	16.7	925	275
24	93.0	132.5	224.3	4.9	17.0	5	16.9	937	272
25	92.8	136.6	226.4	5.7	17.0	5	13.3	724	309
26	93.2	135.9	225.4	5.2	17.0	5	11.9	400	313
27	92.4	138.0	224.1	5.2	18.0	5	11.8	636	341

Table IX

Run "E"	t_1	t_2	T	P	W	θ	w	q_s	q_g
1	91.7	109	238.0	10.4	35	5	6.1	280	259
2	95.0	114.7	237.1	10.5	58	5	13.2	700	474
3	104.7	125.2	236.7	10.1	47	5	10.1	518	400
4	110.3	129.2	237.9	10.6	64	5	6.8	321	502
5	86.6	112.8	236.0	9.5	27.5	5	8.1	400	299
6	88.1	125.9	236.2	10.2	29.5	5	7.2	347	463
7	89.3	128.6	236.6	10.3	29.5	5	7.5	362	480
8	90.0	128.7	236.5	10.1	29.5	5	7.4	360	474
9	101.7	106.4	237.0	10.1	176	5	17.8	996	343
10	108.3	119.7	235.4	9.7	109.5	5	10.3	530	518
11	112.4	123.7	235.7	9.8	103	4	9.7	500	483
12	104.6	121.6	237.3	10.3	50	5	6.9	328	353
13	106.8	124.4	237.0	10.2	53	5	7.7	375	387
14	108.6	126.5	236.8	9.9	57	5	7.1	340	423
15	111.9	129.9	236.5	10.0	57	5	7.5	363	426
16	111.9	121.2	236.5	9.9	100	5	11.2	586	386
17	117.9	127.6	236.4	10.0	134	5	11.1	580	540
18	111.7	121.8	236.4	10.1	39	5	4.8	202	164
19	115.0	126.8	236.9	10.2	44	5	5.7	256	215
20	118.8	129.3	236.2	9.7	77	5	6.9	328	336
21	119.2	131.1	235.9	10.0	50	3	5.9	268	247

Table X

Table of Nomenclature

t_1 .. inlet temperature of fluid, degrees F.

t_2 .. outlet temperature of fluid, degrees F.

T .. steam temperature, degrees F.

P .. steam pressure, lbs./sq. in., gage.

W .. weight of fluid per time θ.

θ .. time of run, minutes.

w .. weight of condensate, ounces.

q_s .. net Btu's given up by steam, (total Btu's —) (radiation loss)

q_g .. Btu's absorbed by the fluid.

u .. mass velocity, lbs./sq. ft./sec.

Q .. Btu's/hour absorbed by the fluid.

ΔT .. over-all temperature drop, $T - \frac{(t_1 + t_2)}{2}$

U .. over-all coefficient of heat transfer, Btu's/hr./sq. ft. of condensing surface/°F over-all temp. drop.

h' .. apparent film coefficient of heat transfer, Btu's/hr./sq. ft. of condensing surface/°F over-all temp. drop.

h .. true film coefficient of heat transfer, Btu's/hr./sq. ft. of inside pipe surface/°F over-all temperature drop.

$\frac{du\rho}{\mu}$ .. Reynold's Number; d .. I. D. of pipe, feet
$u\rho$ .. mass velocity
μ .. absolute viscosity, lbs./(ft.)(sec.)

After conditions of temperature and fluid flow had become fairly constant, readings were taken each minute for five minutes. All runs occupied five minutes except a few, when the time was four or six minutes. The weight rate of flow of the gum was determined by direct weighing on platform scales. The condensate formed during a run was periodically withdrawn in order to keep it from building up into the steam-jacket. Steam temperature in the steam-jacket was measured with mercury thermometers in oil wells in the steam space.

The heat given up by the steam was calculated from the weight of condensate and the latent heat of steam which is 961 Btu/lb. at 5 lbs. gage, and 953 Btu/lb at 10 lbs. gage. Due to the fact that steam at approximately 135 lbs./sq. in. and 97.5% was reduced to the low pressures used, it was assumed that it was either dry or contained a few degrees of superheat. Jakob and Erk (25), however, have shown that superheat has practically no effect on the transfer of heat from condensing steam to metal surfaces. The net amount of heat given up to the gum was determined by subtracting from the total heat, the amount lost by radiation which, by experiment was found to be 745 Btu/hr. at 5 lbs. gage and 1008 Btu/hr. at 10 lbs./gage.

(25) Jakob & Erk; Mech. Eng.; 52; 231 (1930)

The inlet and outlet temperatures are believed to be true average values because the thermometer wells extended almost to the bottom of the pipe thus causing a mixing of the gum. Also, the thermal conductivity of the metal of the well was so great in comparison with that of the fluid that an average value of the cross section temperature was obtained.

Calculated results are shown in Tables XI and XII. The heat absorbed by the gum was calculated from the known weight of gum pumped per unit time, the temperature rise, and the specific heat which by experiment was found to be 0.415 Btu/lb./°F over the temperature range employed. Values of U were calculated as in the runs with water, using the equation

$$Q = U \Delta T A$$

Apparent gum-film coefficients h' were calculated from the equation

$$\frac{1}{U} = r_s + r_p + \frac{1}{h'}$$

Table XI

Calculations and Results

Run "D"	$u\rho$	Q	ΔT	U	h'	h	$(u\rho)\frac{(t_2-t_1)}{(T-t_1)}$	$\frac{du\rho}{\mu}$
15	5.55	756	112.6	3.4	3.4	4.3	0.70	2.29
17	7.22	2240	111.7	10.5	10.6	13.3	1.83	1.75
22	8.86	1930	123.8	8.0	8.1	10.6	1.58	1.51
23	9.44	3300	114.9	14.7	14.9	18.7	2.73	3.61
24	9.44	3260	111.5	14.9	15.1	18.9	2.77	4.31
25	9.44	3710	111.7	17.0	17.3	21.6	3.09	5.11
26	9.44	3610	110.8	16.6	16.9	21.3	3.05	5.12
27	10.00	4090	108.9	19.2	19.5	24.5	3.46	5.55
2	12.50	820	109.1	3.8	3.9	4.8	0.81	6.26
18	14.45	4240	121.9	18.0	18.3	22.9	3.42	2.81
19	15.01	4500	118.4	19.4	19.7	24.7	3.72	4.11
20	15.01	4690	119.2	20.1	20.4	25.5	3.83	3.90
3	20.83	900	108.6	4.2	4.2	5.8	0.90	11.3
8	22.16	3650	128.4	14.5	14.7	18.4	2.95	2.2
1	22.45	2700	108.9	12.7	12.8	16.1	2.60	12.2
9	23.41	5790	122.4	24.2	24.7	31.0	4.81	4.88
10	25.82	7500	119.1	32.2	33.1	41.5	6.19	6.71
13	30.56	8680	115.1	38.5	39.8	49.9	7.26	11.0
4	32.63	3360	122.4	14.0	14.1	17.7	2.93	5.91
21	33.89	3960	118.9	17.0	17.3	21.6	3.20	8.81
12	40.56	6000	116.7	26.5	26.9	33.7	5.35	12.9
7	42.15	4600	101.5	23.1	23.6	29.6	4.78	49.4
11	45.00	3760	121.2	15.8	16.0	20.1	3.33	10.5
6	65.00	5000	102.3	25.0	25.5	32.1	5.25	65.0
16	72.50	4310	111.2	19.8	20.1	25.3	4.18	34.3
5	78.07	4120	113.4	18.6	18.9	23.7	3.96	28.2
14	84.90	4580	114.7	20.6	21.0	26.3	4.33	30.7

Table XII

Calculations and Results

Run "E"	$u\rho$	Q	ΔT	U	h'	h	$(u\rho)\frac{(t_2 - t_1)}{(T - t_1)}$	$\frac{du\rho}{\mu}$
5	15.28	3590	136.3	14.8	15.0	18.8	2.68	2.56
6	16.39	5560	129.2	22.0	22.4	28.1	4.18	4.89
7	16.39	5760	127.7	23.3	23.8	29.8	4.37	5.76
8	16.39	5690	127.1	22.8	23.3	29.2	4.32	5.84
1	19.45	3110	137.4	11.6	11.7	14.7	2.36	3.49
18	21.67	1970	119.7	8.4	8.5	10.6	1.75	13.8
19	24.26	2580	116.0	11.3	11.4	14.3	2.37	22.0
3	26.00	4800	121.8	20.1	20.5	25.6	4.05	14.2
12	27.77	4240	124.2	17.4	17.7	22.1	3.55	13.1
13	29.45	4640	121.4	19.5	19.8	24.9	3.97	17.4
14	31.68	5080	119.7	21.7	22.1	27.7	4.43	20.6
15	31.68	5110	115.6	22.6	23.1	28.9	4.58	28.4
2	32.20	5690	132.2	22.0	22.4	28.1	4.46	7.8
4	35.55	6020	118.1	26.0	26.6	33.3	5.26	28.9
20	42.75	4060	112.2	18.5	18.8	23.6	3.82	48.4
21	46.32	4940	110.8	22.8	23.3	29.2	4.75	57.4
16	55.57	4630	120.0	19.7	20.0	25.1	4.15	35.3
10	60.82	6220	116.4	27.3	28.0	35.1	5.45	48.0
11	71.50	7240	117.6	31.4	32.3	40.5	6.55	50.3
17	74.45	6480	113.7	29.1	29.8	37.4	6.10	77.5
9	97.80	4120	132.9	15.8	16.5	20.7	3.39	25.5

where $r_s + r_p$ is the resistance of the steam-film and pipe as determined in the calibration runs with water. Although the steam pressure was not the same as in the case with water, the steam-film coefficient (and, of course, the steam-film resistance) remained unchanged, because, as pointed out by Webster (26), the steam-film coefficient is apparently independent of the steam pressure.

A sample calculation will illustrate the method used in computing h.

Run 18, Series D

$$Q = M(t_2 - t_1)c$$

Q - Btu's/hr. absorbed by gum

M - lbs./hr.

t_1 - inlet temp. °F

t_2 - outlet temp. °F

c - specific heat = 0.415 Btu/lb/ F.

$$Q = 312(117.7 - 85)0.415$$

$$Q = 4240 \text{ Btu/hr.}$$

(26) Webster; Inst. Eng. Shipb. Scot.; 57; 58 (1913)

$$U = \frac{Q}{\Delta TA}$$

U - over-all coefficient of heat transfer

ΔT - over-all temperature drop, steam to arithmetical mean temperature of gum. (The log mean temp. of the gum is not used because the temp. distribution in the fluid and the local doefficients of heat transfer vary along the pipe.)

A - outside area of heated section of pipe 1.958 sq. ft.

$$U = \frac{4240}{121.9 \times 1.958}$$

$$U = 18.0 \text{ Btu/hr./sq. ft./}^{\circ}\text{F}$$

$$\frac{1}{U} = r_s + r_p + \frac{1}{h'}$$

h' - apparent gum-film coefficient based on one sq. ft. of outside pipe surface.

$r_s + r_p =$ 0.00086 (see section on water runs.)

$$\frac{1}{18} = 0.00086 + \frac{1}{h'}$$

$h' = 18.28$ Btu/hr./sq. ft. of outside area/ $^{\circ}$F overall temp. drop.

$$h = \frac{D'}{D} h'$$

h - gum-film coefficient based on one sq. ft of inside area.

D' - outside diameter of pipe, inches.

D - inside diameter of pipe, inches

$$h = \frac{1.315}{1.049} \; 18.24$$

$$h = 22.9 \text{ Btu/hr./sq. ft./}^{\circ}\text{F}$$

Discussion of Results

The data of the two series of runs are well correlated by plotting on ordinary rectangular coordinates, values of the film-coefficient h as ordinates against the product of mass velocity and the ratio of temperature rise to initial temperature differences; i.e. h vs $(u\ )\frac{(t_2 - t_1)}{(T - t_1)}$.

Figure 5 is such a plot. From Tables XI and XII and Fig. 5 it is apparent that the increased steam temperature used in Series E is without effect on the film-coefficient of heat transfer of the gum. The advantage of using higher steam temperatures lies in the fact that greater temperature drops, ΔT, are obtained and consequently more Btu's per unit time are transferred.

The equation of the straight line in Fig. 5 is

$$h = 6.41(u\rho)\frac{(t_2 - t_1)}{(T - t_1)}$$

It is believed that this euqation for h is valid only in the region of stream-line flow. Due to the high viscosity of the gum turbulent flow would be obtained only at unusually high temperatures. At such temperatures the turpentine of the gum would distill off thus changing the composition of the gum and consequently, changing the values of the gum film-coefficient.

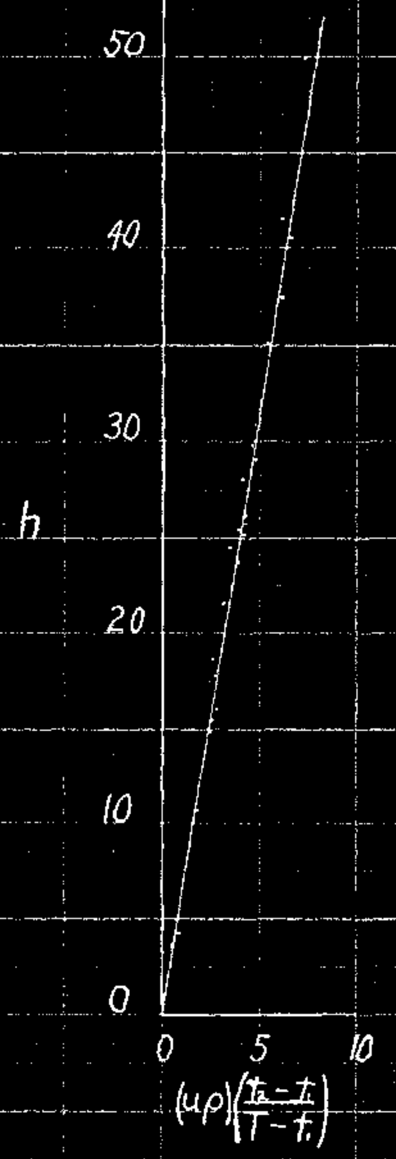

Figure 5

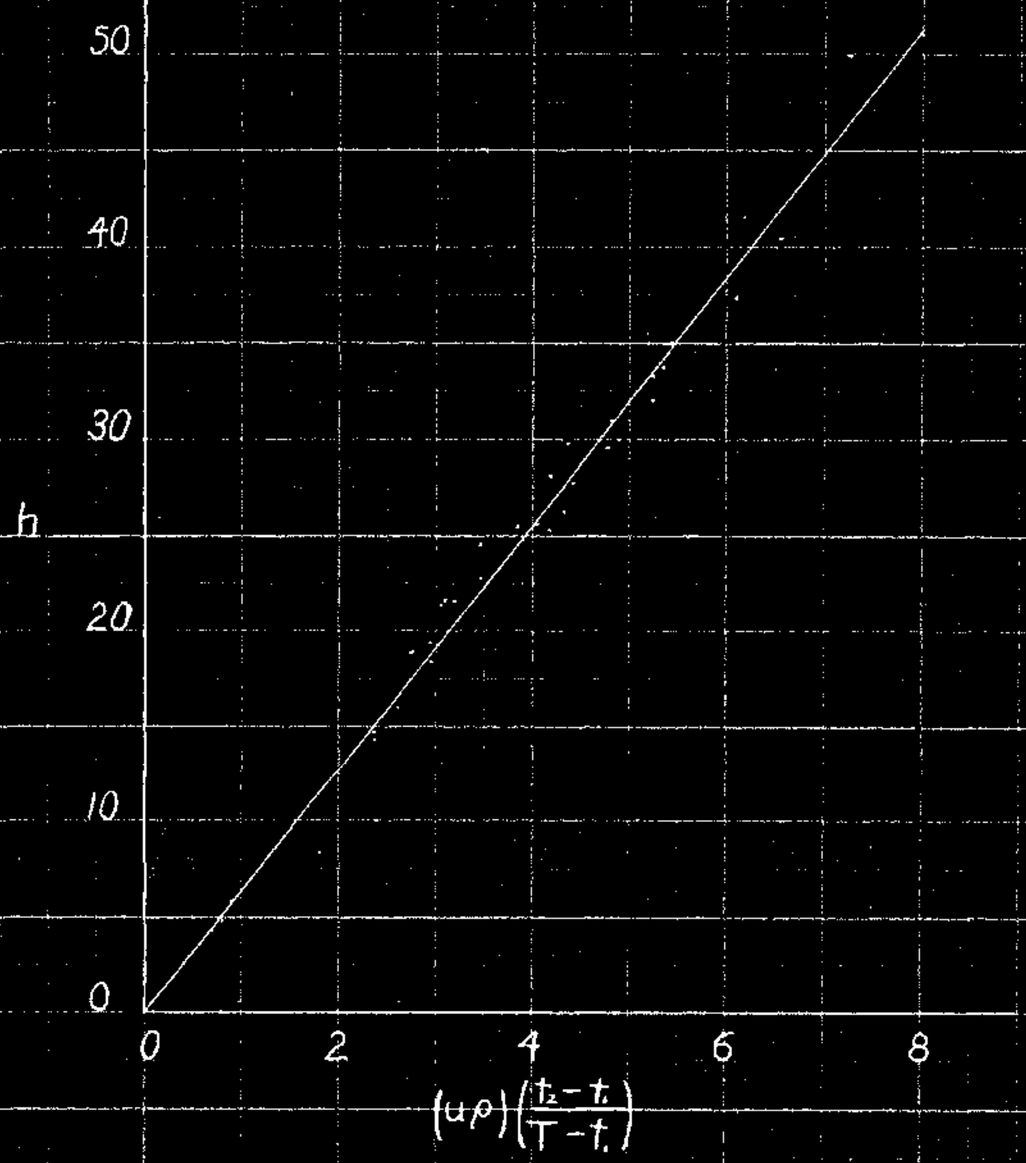

Figure 5

It is hoped that the data obtained during this investigation will be of value in the design and operation of turpentine distillation equipment.